PART 7 – HEAT EJECTION SYSTEMS

Section

GENERAL HEAT EJECTION CONSIDERATIONS

Heat ejection system parameters are based largely on the transmitter power, efficiency, operating temperature and spacecraft thermal environment.

Thermal control system requirements imposed by the communication system are presented in this section. The communication system characteristics which determine the thermal control requirements are largely the output power, efficiency, and operating temperature of the transmitting source. In addition, the thermal control system burden is also influenced by the mission thermal environment and the spacecraft configuration.

Since operation of the transmitter will not be continuous throughout the mission, the steady state thermal control of the spacecraft is dependent upon the rate of heat or power ejected. Thus, essentially all the heat produced by intermittent operation of the transmitter must be ejected from the spacecraft. The power to be ejected, W, is then

$$W = P_T \frac{1 - k_e}{k_e}$$

where

P_T = transmitter output power

k_e = transmitter efficiency

This approximation of ejected heat burden is totally conservative since it is assumed that none of the heat ejected by the transmitter is put to effective use in thermal control.

The system parameters of most significance in determining the radiator burden for a given transmitted power are the operating temperature of the transmitting source and its efficiency. Operating temperatures range from less than 40° C for some laser sources to 200 to 250° C for TWT microwave sources. The efficiencies vary even more widely. If the transmitter power and efficiency have been specified and if its operating temperature is known, the associated radiator weight, area, and cost are essentially determined.

TRANSMITTER SOURCE CHARACTERISTICS

Microwave and laser transmitting sources vary markedly in their sensitivity to operating temperature, their efficiency and their allowable operating temperature.

To gain an appreciation for the range of parameters into which a thermal head radiator must match, the power amplifiers for microwave links and laser links are considered below.

Microwave Sources. A microwave source of prime interest for long space missions is the traveling-wave amplifier tube (TWT). In a TWT the greatest heat is generated at the collector surface. These parts may reach temperatures as high as 200 to 250°C in present long life tubes. For lower power levels (less than 100 to 200 watts output) it is customary to conductively cool the collector by thermally connecting it to a heat sink. The heat sink in turn conducts the heat to an external radiating surface. Higher power tubes are customarily cooled by flowing a coolant fluid through integral passages in the collector and other critical parts. The upper limit in outlet fluid temperature is imposed by the collector temperature limitations although it is typically somewhat lower as a result of temperature drops in other parts of the internal tube coolant circuit.

For power levels beyond 1 kw, TWTs are generally built in a different configuration from that used at lower powers. The high power configuration uses a cavity resonator which requires a solenoid to provide the necessary magnetic field. The solenoid must be cooled as well in this case. With modern high temperature insulating materials the solenoid operating temperatures may be comparable with the collector temperature.

Traveling wave tubes operate at efficiencies as high as 30 percent including power supply losses. For high power tubes, the solenoid cooling requirement is reflected in the efficiency.

Optical Transmitting Sources. The two optical sources of primary interest are the CO_2 laser (10.6 μ wavelength) and the Argon laser (0.5145 μ wavelength). They differ drastically in operating efficiency and operating temperature requirements.

The CO_2 laser operates at efficiencies as high as 15 percent. To achieve this high efficiency, the gas mixture must be kept at temperatures of 20°C or less. Efficiency drops off rapidly at higher temperatures. For a typical low power device, output was reduced from 1.5 watts at 20°C to 0.7 watt at 60°C and 0.25 watt at 100°C. To maintain the required temperatures, most laboratory versions use water as a coolant, flowing it between the walls of the discharge tube and an outer concentric jacket. For the higher power levels envisioned for space transmitting sources, an active fluid cooling system is virtually a necessity.

The Argon laser is characterized by efficiencies of the order of 0.1 percent or less. The very large fraction of input power which must be rejected as a result of this inefficiency demands liquid cooling for all power levels under consideration. Efficiency is not a critical function of the operating temperature as is the case with the CO_2 laser.

The upper limit in operating temperature is imposed by the limits for safe operation of the solenoid which surrounds the discharge tube and provides the pumping magnetic field. The pumping solenoid generates a large amount of heat and both it and the discharge tube must be cooled. The most effective way to achieve this is to flow the coolant fluid through the annular passage between them. Maximum operating temperatures imposed by solenoid temperature as limited by modern high temperature insulating materials may be as high as 100 to 150°C.

SUMMARY

Heat ejection design is based upon ejecting heat into space.

Spacecraft heat ejection systems are able to eject heat into space by radiation. The amount of heat that is ejected depends upon several factors. These include the temperature of the surface ejecting heat, T; the surface emissivity, ϵ ; a constant, which is the Stefon-Baltzmann's constant, $\sigma = 5.7 \times 10^{-12}$ watts/cm^2 $^\circ$K; the temperature of the sink, Ts, (0°K for free space, somewhat higher when near a planet.); and heat loading, which is predominantly from the sun. This loading depends upon the solar illumination, H; the absorbtivity of the surface, α_s; and the angle at which the sun's range strike the surface, θ. Expressed in a single equation, the heat ejected in watts/cm^2 is:

$$Q = \epsilon\sigma (T^4 - T_s^4) - \alpha_s H \cos\theta$$

Clearly there are many variables in this equation. Ranges for these are given with sample calculations in the material which follows.

HEAT EJECTION SYSTEMS

Heat Ejection Elements

HEAT EJECTION ELEMENTS

TYPES OF HEAT EJECTION SYSTEMS

An active heat exchanger is one in which the heat is conveyed from the heat source to the radiating element by a moving coolant or moving mechanical parts, while a passive heat exchanger has no moving parts.

Heat ejection systems may be classified as active or passive (see the figure). In the most general sense, an active system is one which embodies moving parts (e.g., a coolant fluid or a thermal switch) while a passive system does not. In typical active systems heat is conveyed to the radiating surface by first transferring it to a fluid medium which is then physically transported to the radiator where its heat is ejected. In a passive system heat is conveyed to a radiating surface and dissipated from it by purely static processes.

Passive Heat Ejection

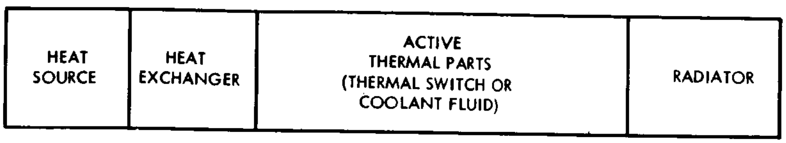
Passive heat ejection systems are preferable when they can meet the requirements because of their extreme simplicity and concomitant lighter weight, lower cost, and higher reliability. They consist merely of a conducting path between the heat source and an external radiating surface, often a part of the spacecraft structure, having a highly emissive surface coating with low solar absorptivity. The limitation on their utility is almost always excessive temperature gradients in the conducting path as a result of thermal resistance.

Passive and active heat ejection systems differ in the method of conducting the heat from one point to another but the same considerations apply to the actual radiators of heat. The heat radiators are considered in some detail in later topics of this section.

Active Heat Ejection

Active heat ejection systems generally consist of a heat exchanger to transfer heat from the transmitting source to the cooling fluid, the necessary plumbing to convey the fluid to the radiator, and the radiator itself. Of these, the radiator proper is the major contributor to the thermal control system cost, weight, and area burdens. The heat exchanger at the transmitting source is an integral part of the source and is characteristic of it. The burdens associated with transferring heat from the source to the cooling system are thus included in the transmitting source burdens and cannot meaningfully be divorced from them. The remaining system components – plumbing, pumps, controls and the coolant itself – are of less significance than the radiator with respect to cost, weight, and volume. They are, in any event, so peculiar to a specific vehicle and communication system configuration as to preclude meaningful treatment here.

Both condensing and non-condensing active heat ejection systems will be discussed in subsequent topics. Condensing (two phase) systems are most applicable to dynamic power systems and so are included as a matter of general interest. Non-condensing (single phase) systems appear more applicable to cooling transmitting sources since boiling of the coolant fluid in condensing systems introduces vapor pockets and would lead to local hot spots in critical areas.



(a) ACTIVE SYSTEM

(b) PASSIVE SYSTEM

Heat Ejection System Constituant Parts

HEAT PIPE[1]

The unique feature of a heat pump is the use of capillary action to "pump" a fluid.

The heat pipe is essentially a closed, evacuated chamber whose inside walls are lined with a capillary structure, or wick, that is saturated with a volatile fluid (see the figure). The operation of a heat pipe combines two familiar principles of physics: vapor heat transfer and capillary action. Vapor heat transfer is responsible for transporting the heat energy from the evaporator section at one end of the pipe to the condenser section at the other end. What distinguishes the heat pipe is that the heat pipe capillary action is responsible for returning the condensed working fluid back to the evaporator section to complete the cycle.

The function of the working fluid within the heat pipe is to absorb the heat energy received at the evaporator section (by the latent heat at evaporation), transport it through the pipe and release this energy at the condenser end.

As the working fluid vaporizes, the pressure at the evaporator end of the pipe increases. The vapor pressure sets up a pressure difference between the ends of the pipe, and this pressure difference causes the vapor, and thus the heat energy, to move toward the condenser section. When the vapor arrives at the condenser section, it encounters a temperature lower than that of the evaporator. As a consequence the vapor turns back to a liquid and thereby releases the thermal energy stored in its heat of vaporization.

As the fluid condenses, the vapor pressure created by the molecules decreases, so that the necessary pressure difference for continual vapor heat flow is maintained.

It is important to note that the vaporized fluid stores heat energy at the temperature at which the vapor was created, and that it will retain the energy at that temperature until it meets a colder surface. The result is that the the temperature along the entire length of the heat pipe tends to remain constant. It is this tendency to resist any difference in temperature within the heat pipe that is responsible for the device's high thermal conductance.

One of the requirements of any self-contained vapor heat-transfer system is a means of returning the condensed liquid to the evaporator to replenish the supply. In older systems this return was accomplished either by gravity or by a pump. The heat pipe, on the other hand, can operate against gravity and without a second external energy source. This is accomplished by capillary action within the wick that connects the condenser to the evaporator. The capillary action "pumps" the cooled fluid back to the heat source.

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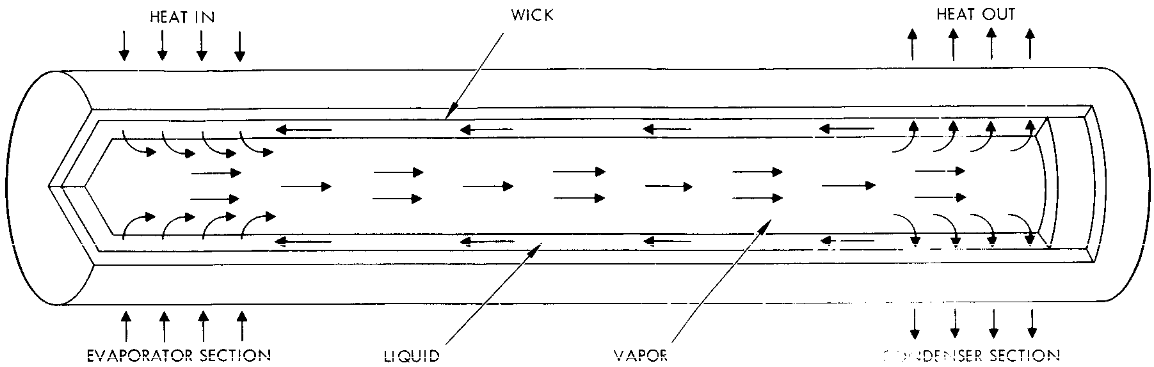
[1] G. Y. Eastman "The Heat Pipe", Scientific American, May 1968, pp 38-46.


HEAT IN
WICK
HEAT OUT
EVAPORATOR SECTION
LIQUID
VAPOR
SECTION

Heat Pipe

USEFUL HEAT PIPE PROPERTIES

Five useful properties of heat pumps, as an example of an active heat radiator element, are described.

Heat Pipe Properties

There are five properties of the heat pipe that deserve special mention because they serve to define the areas in which practical applications are to be found for the device. These are listed briefly below.

First, active devices that operate on the principle of active vapor heat transfer can have several thousand times the heat-transfer capacity of the best passive metallic conductors, such as silver and copper.

A second property of the heat pipe is called "temperature flattening." There are many heat-transfer applications in which a uniform temperature over a large surface area is required. Without the heat pipe special care must be taken to ensure a uniform temperature of the heat source. A heat pipe, however, can be coupled to a nonuniform heat-source to produce a uniform temperature at the output, regardless of the point-to-point variations of the heat source.

Third, the evaporation and condensation functions of a heat pipe are essentially independent operations connected only by the streams of vapor and liquid in the pipe. The patterns and area of evaporation and condensation are independent. Thus the process occurring at one end of the pipe can take place uniformly or nonuniformly, over a large or a small surface area, without significantly influencing what is going on at the other end.

A fourth property of the heat pipe is that it makes it possible to separate the heat source from the heat sink.

Fifth, the heat pipe can also be operated so that the thermal power and/or the temperature at which the power is delivered to the intended heat sink can be held constant in spite of large variations in the power input to the heat pipe.

Heat Pipe Implementation

Heat pipes have been made to operate at various temperatures spanning the range from below freezing to over 3,600 degrees F. The power transferred ranges from a few watts to more than 17,000 watts. Working fluids have included methanol, acetone, water, fluoridated hydrocarbons, mercury, indium, cesium, potassium, sodium, lithium, lead, bismuth and a range of inorganic salts. The containment vessels have been made of glass, ceramic, copper, stainless steel, nickel, tungsten, molybdenum, tantalum and various alloys. The wicks or capillary structures have included sintered porous matrixes, woven mesh, fiber glass, longitudinal slots and combinations of these structures in various geometries. In physical size heat pipes have ranged from a quarter of an inch to more than six inches in diameter and up to several feet in length. Moreover, heat pipes can be designed in almost any configuration.

An operating life in excess of 10,000 hours without failure or detectable degradation has been achieved with a range of fluid-container systems. The longest of these tests has currently passed 16,000 hours at 1,100 degrees F., using potassium as the working fluid in a nickel containment vessel.

HEAT EJECTION SYSTEMS

Heat Ejection Elements – Radiators

RADIANT HEAT TRANSFER FROM A FLAT SURFACE

The equations describing radiant heat transfer from a flat surface are given and pertainant curves are drawn.

For typical spacecraft fin and tube radiators the controlling thermal resistance is conduction and radiation in the radiator fin. Therefore the fin design is a chief concern for the preliminary designer, while heat transfer from the working fluid to the fin is a second-order problem.

Radiant heat transfer from a flat surface at temperature, T, to a sink at absolute zero is described by the Stefan-Boltzmann equation:

$$Q = \epsilon \sigma T^4 \tag{1}$$

where

Q = radiative power (watts/cm^2)

ϵ = surface emissivity

σ = Stefan-Boltzmann constant = 5.7×10^{-12} watts/cm^2 °K

T = radiating surface temperature (°K)

For a non-zero sink temperature, this expression becomes

$$Q = \sigma \epsilon (T^4 - T_s^4) \tag{2}$$

where

T_s = sink temperature (°K)

If solar illumination is incident on the radiator, it must also be ejected, reducing the effective radiative heat flux (i. e., dissipation of heat produced by an on-board source) to

$$Q = \epsilon \sigma (T^4 - T_s^4) - \alpha_s H \cos \theta \tag{3}$$

where

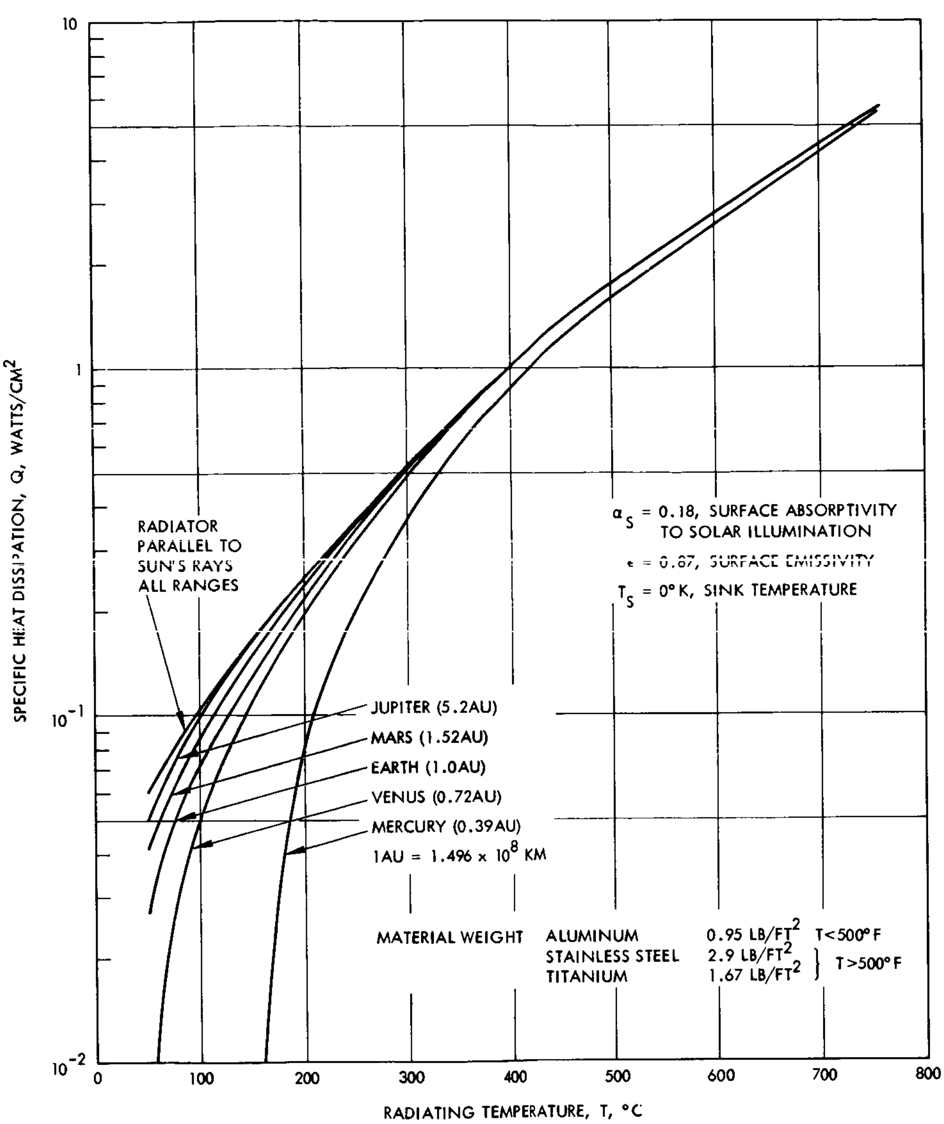
α_s = surface absorptivity to solar illumination

H = solar illumination intensity (0.138 watts/cm^2 at 1 A.U.)

θ = angle between the incident solar illumination and the normal to the radiator surface

ϵ = surface emissivity

Equation (3) is plotted in the figure indicating the dependance of the heat radiation upon solar radiation. As may be seen at ranges greater than 1 to 1.5 AU from the sun, the solar effect does not cause a major variation in the radiator design.



Specific Heat Dissipation Capacity Versus Temperature of Radiators Formed to Direct Sunlight at Various Solar Distances

RADIATOR FIN EFFECTIVENESS

The fin effectiveness compares the actual radiative effectiveness of a radiator with the maximum effectiveness.

A quantity termed the fin effectiveness is introduced to assist in the evaluation of the performance of a finned radiator. It is defined as the ratio of the heat ejected by the fin to that which would be ejected if the entire fin were maintained at the base temperature. The fin effectiveness, η, may be included in the equation defining the specific heat radiation Q, as follows.

$$Q = \epsilon \sigma \eta (T^4 - T_s^{\,4}) - \alpha_a H \cos \theta$$

where

α_s = surface absorptivity to solar illumination

H = solar illumination intensity (0.138 watts/cm^2 at 1 AU)

θ = angle between the incident solar illumination and the normal to the radiator surface

ϵ = surface emissivity

σ = Stefan-Boltzmann constant = 5.7×10^{-12} watts/cm^2 ^oK

T = radiating surface temperature (^oK)

T_s = sink temperature (^oK)

Expressed mathematically:

$$\eta = \frac{\int_o^{\frac{B}{2}} T_x^{\,4}\, dx}{\frac{B}{2}\, T^4} \tag{4}$$

where

η = fin effectiveness

B = tube spacing

T_x = temperature at a point on the fin

x = distance along fin

T = fin base temperature

This equation was derived by Coombs[1] et al., and was solved numerically on an IBM-704 computer. The results are given in the figure as a function of the dimensionless radiation modulus M_r defined as:

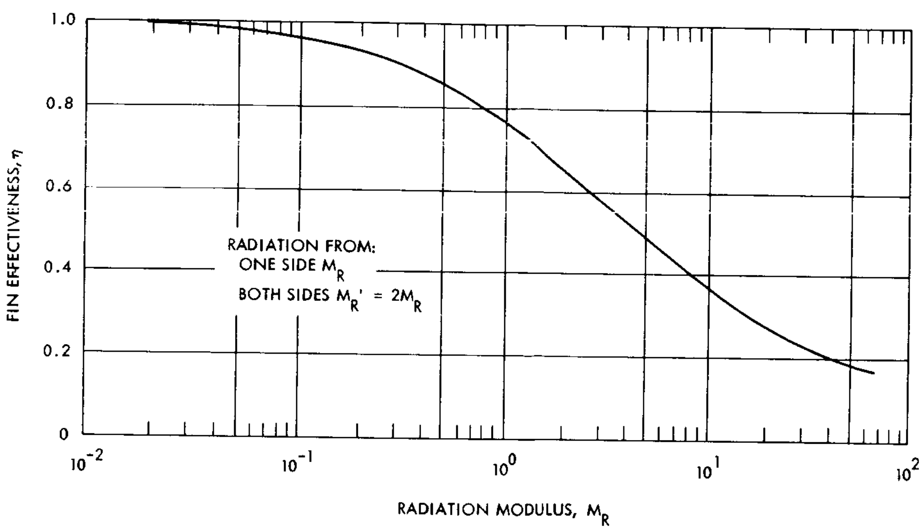
$$M_r = \frac{B^2 \epsilon \sigma T^3}{kt} \tag{5}$$

where

k = conductivity of fin material

t = fin thickness

By using the curve in the figure the fin effectiveness may be evaluated for a given material, geometry, and base temperature, T.



Fin Effectiveness versus Radiation Modulus

[1] M. G. Coombs, R. A. Stone, and T. Kapus, "The SNAP 2 Radiative Condenser Analysis," NAA-SR-5317, July 1960.

RADIATOR AREA REQUIREMENTS — CONDENSING SYSTEMS

The specific heat radiated from a surface depends critically on the radiator orientation and emissivity especially at lower radiating temperatures.

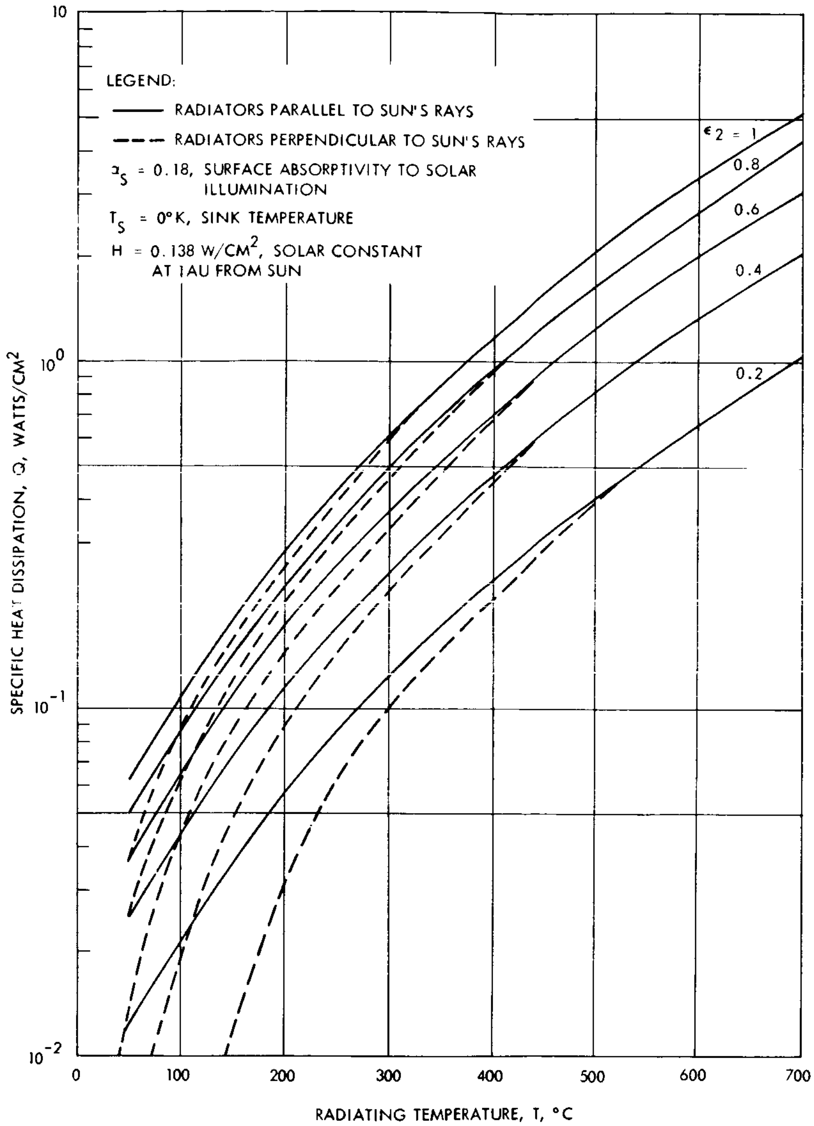
For condensing radiators, the tube temperature remains constant until the fluid is completely condensed, as long as the static pressure drop is kept small. This follows since the condensate and condensing vapor are always in thermal equilibrium. If this condition is met, the area requirements for the condensing portion of the radiator can be obtained from the total power which must be dissipated and the radiative power per unit area, Q:

$$Q = \epsilon \sigma \eta \left(T^4 - T_s^4\right) - \alpha_s H \cos \theta \qquad (1)$$

where

- Q = radiative power (watts/cm^2)
- ϵ = surface emissivity
- σ = Stefan-Boltzmann constant = 5.7×10^{-12} watts/cm^2 °K
- T = radiating surface temperature (°K)
- T_s = sink temperature (°K)
- η = fin effectiveness
- α_s = surface absorptivity to solar illumination
- H = solar illumination intensity (0.138 watts/cm^2 at 1 AU)
- θ = angle between the incident solar illumination and the normal to the radiator surface

Equation 1 has been plotted in the figure as a function of radiating temperature with the product of surface emissivity, ϵ, and fin effectiviness, η, as a parameter. Two sets of curves are shown in the figure. One set corresponds to the specific heat dissipation when the radiating surface is parallel to the sun's rays (best case) and the second set corresponds to the sun's rays being normal to the radiating surface (worst case). As may be seen from the figure the orientation of the radiating area becomes critical at lower radiating temperatures and lower effective emissivities.



Specific Heat Dissipation Capacity Versus Radiator Temperature

RADIATOR AREA REQUIREMENTS — NON CONDENSING SYSTEMS

A non-condensing radiator has a temperature gradient along the radiator length. This causes this type of radiator to be less efficient than a condensing radiator.

In non-condensing systems the radiant heat rejection is accompanied by a sensible heat loss of the fluid. The temperature decrease of the fluid results in temperature gradients both perpendicular and parallel to the direction of fluid flow. This complicates the analysis, but by combining the model of the condensing (constant temperature) fin with that of a radiator which experiences a coolant temperature drop, an expression can be derived to give the area requirements for the tube-fin configuration.[1] The result is given by:

$$Q = \eta \sigma \epsilon \left[\frac{3\, T_{in}^{3}\, T_{out}^{3}}{T_{in}^{2} + T_{in} T_{out} + T_{out}^{2}} \right] \qquad (1)$$

where

Q = radiative power (watts/cm^2)

η = fin effectiveness

σ = Stefan-Boltzmann constant = 5.7×10^{-12} watts/cm^2 $^\circ$K

ϵ = surface emissivity

T_{in} = fluid temperature into radiator ($^\circ$K)

T_{out} = fluid temperature out of radiator ($^\circ$K)

The radiative power, Q, is plotted in the figure as a function of temperature drop of the non-condensing fluid across the radiator length. The input temperature to the radiator is used as a parameter. The effectiveness is shown in the figure to decrease with increasing temperature drop along the radiator length.

Equation 1 has been written not considering the heat loading of the sun or the sink temperature. If these factors are included the new equation is as follows:

$$Q = \eta \sigma \epsilon \left(T_{eff}^{4} - T_{s}^{4} \right) - \alpha s H \cos \theta$$

where

$$T_{eff} = \left[\frac{3\, T_{in}^{3}\, T_{out}^{3}}{T_{in}^{2} + T_{in} T_{out} + T_{out}^{2}} \right]^{1/4}$$

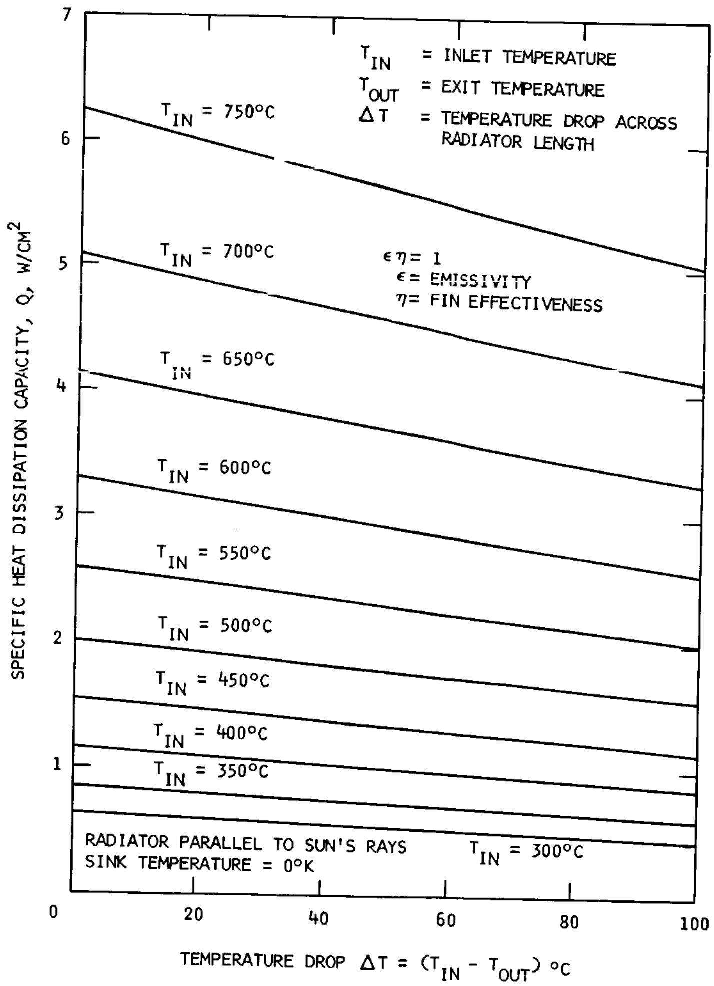
[1] Energy Conversion Systems Reference Handbook, Volume X, Reactor System Design, Electro-Optical Systems, September 1960.

T_s = sink temperature

α_s = surface absorptivity to solar radiation

H = solar constant (0.138 w/cm^2 at IAU)

θ = angle between the incident solar illumination and the normal to the radiator surface



Area Requirements for Non-Condensing Radiator

HEAT EJECTION SYSTEMS

Weight and Cost Burdens

WEIGHT AND COST BURDENS

RADIATOR WEIGHT AND COST VARIATIONS

Radiator weights and cost vary over wide ranges depending upon the specific heat dissipation.

According to AiResearch Corporation,[1] low temperature radiator specific weight, assuming aluminum construction and structural rigidity as required for radiator areas greater than 50 ft^2, is approximately 0.95 lb/ft^2. For smaller radiator areas, depending on the amount of structural rigidity required, the specific weight may be as low as 0.045 lb/ft^2. Radiator heat dissipation capability versus weight is plotted in Figure A based on 0.95 lb/ft^2. Typical costs as quoted by the same source indicate development costs of $50,000, exclusive of environmental testing, for one 10 to 30 ft^2 space qualified radiator. For production of a number of identical radiators with the above development cost amortized over five units, an approximate functional relationship between radiator cost, C_H, and area, A, of

$$C_H = \$13,750 + 75\ A \tag{1}$$

can be inferred. For large production runs, with the development cost amortized over one hundred units, the radiator cost is reduced to

$$C_H = \$2,750 + 25\ A \tag{2}$$

Radiator heat dissipation capability versus cost for a five unit production run is plotted in Figure B.

In addition to the heat dissipation capacity of the radiator, three significant factors strongly affect the required cost and weight of the radiator. These factors are: 1) the radiating temperature of the heat radiator (note that this can vary with the type of power amplifier on the spacecraft.); 2). the sink temperature, T_s, in to which the radiator radiates; and 3) the aspect of the radiator to the heat energy from the sun. These are, of course, all factors in determining the specific heat dissipation, Q. Figures A and B indicate typical variations in cost and weight with typical values of emissivity, fin effectiveness, and surface absorptivity.

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1. Private Communication, AiResearch Corporation

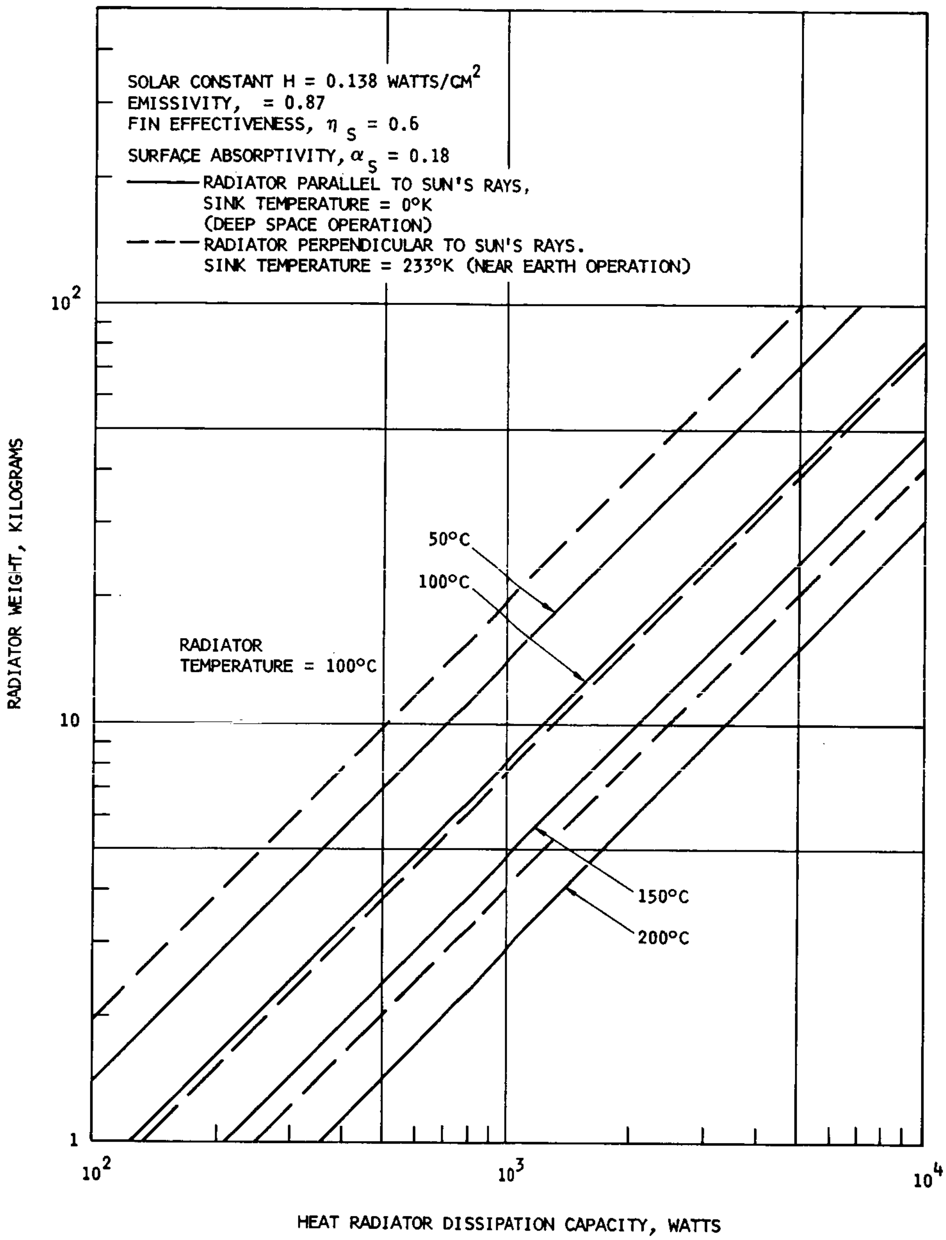

Figure A. Fin and Tube Radiator Weight, W_H (Kilograms), Versus Heat Dissipation Capacity at Various Radiator Temperatures

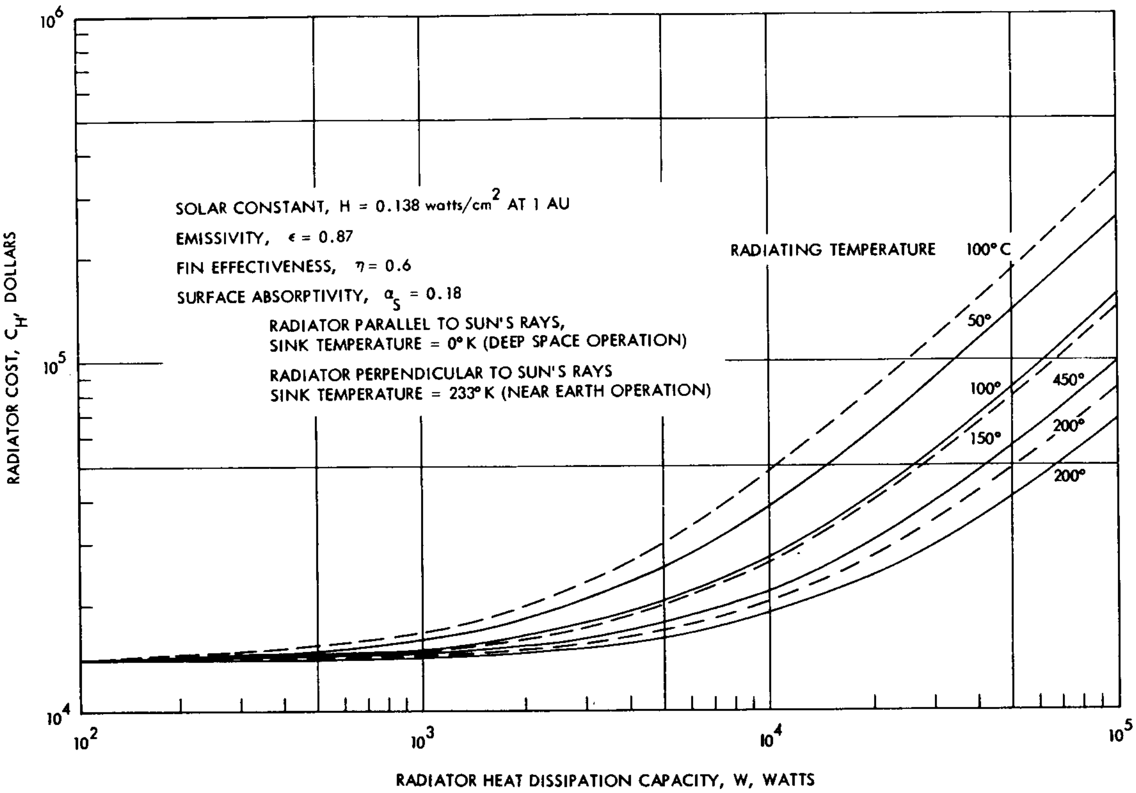

Figure B. Fin and Tube Radiator Cost, C_H, Versus Heat Dissipation Capacity at Various Radiator Temperatures

WEIGHT AND COST BURDEN CONSTANTS

Weight and cost burden for the communications system methodology are derived from the previous topic.

An overall goal of this final report is to provide a means for comparing various communication systems on the basis of cost and weight. To accomplish this it is necessary to express each component in terms of its pertained parameter(s) and weight and cost. This topic lists these relationships for the heat exchanger/radiator.

The fabrication cost and weight of a heat exchanger are proportional to the power which is to be radiated. The heat exchanger weight, W_H, given in the nomenclature of the communications system methodology is:

$$W_H = K_X \left(\frac{1 - k_e}{k_e}\right) P_T + W_{KH}$$

and the heat exchanger fabrication cost, C_H, is

$$C_H = K_H \left(\frac{1 - k_e}{k_e}\right) P_T + C_{KH}$$

where

W_{KH} = transmitter heat exchanger weight independent of the radiated heat

C_{KH} = heat exchanger fabrication cost independent of power dissipation

K_X = constant relating heat exchanger weight to power dissipation

K_H = constant relating heat exchanger fabrication cost to power dissipation

k_e = power efficiency, from the prime power source to the output power

P_T = transmitted power

It is the purpose of this topic to tabulate the values of K_X, W_{KH}, K_H, and C_{KH}. These may be derived from the previous topic and are given in Tables A and B. As may be noted from these tables the burden values are strongly dependent upon the radiating temperature and upon the sun's heat load on the radiator. Thus different values of these burdens are associated with different temperature maintainance requirements (e.g., CO_2 laser requires relatively low operating temperature as compared to a TWT which can operate at a much higher temperature.)

Table A. Values of the Heat Exchanger Weight Burdens[1]

Radiating Temperature, °C	W_{KH} Pounds	W_{KH} Kilograms	K_X Pounds/Watt	K_X Kilogram/Watt
Radiator parallel to sun's rays, sink temperature = 0°K (Deep space operation)				
50	0	0	0.031	0.014
100	0	0	0.0175	0.008
150	0	0	0.011	0.0049
200	0	0	0.0068	0.00311
Radiator perpendicular to sun's rays, sink temperature = -40°C (Near earth operation)				
50	0	0	*	*
100	0	0	0.042	0.019
150	0	0	0.0165	0.0075
200	0	0	0.0088	0.004

*Input heat exceeds radiated heat.

Table B. Values of the Heat Exchanger Cost Burdens[1]

Radiating Temperature, °C	C_{KH} Dollars	K_H Dollars/Watt
Radiator parallel to the sun's rays, sink temperature = 0°K (Deep space operation)		
50	13,750	2.5
100	13,750	1.4
150	13.750	0.84
200	13.750	0.54
Radiator perpendicular to the sun's rays, sink temperature = -40°K (Near earth operation)		
50	13.750	*
100	13.750	3.36
150	13.750	1.31
200	13.750	0.70

*Input heat exceeds radiated heat.

* Heat radiator parameters are emissivity, $e = 0.87$; surface absorptivity, $\alpha_s = 0.18$; fin effectivity, $\eta = 0.6$; solar constant $H = 0.138$ watt/cm^2.

NASA-Langley, 1970 —— 7 CR-1688